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Article

Study of Brake Pad Shim Modification to Improve Stability Against High Frequency Squeal Noise by Finite Element Analysis

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Abstract. This research studying on brake pad shim design with 4 configurations to improve the brake squeal noise phenomenon for high frequency noise in the range of 4-16 kHz. Various shims were designed with different configurations to increase structural damping and avoid instabilities in a brake system, which arise from friction drawing the vibration modes to coalesce between brake disc and brake pads. Then, the suspect brake module was tested in the laboratory using a dynamometer machine to confirm brake frequency noise parameters and conditions. The numerical models including brake disc, brake pad and brake pad shim were created using finite elements software and the unstable modes analysed for negative damping and positive real part values with the Complex Eigenvalue Analysis (CEA) technique. The simulation result showed that the instability of the brake system comes from mode coupling of the brake disc and brake pads in the out-of-plane modes (11ND) and (2 ND), respectively. The brake pads shim design1, design2 and design3 are component which goes in between the calipers and brake pads, were able to avoid high frequency brake squeal but make the noise move toward the direction of the lower frequency. The brake pad shim design4 is the good structure modification to avoid high frequency brake squeal and low frequency.

Keywords: Modal coupling, high frequency brake squeal noise, finite element analysis, complex eigenvalue analysis, modal testing.

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1. Introduction

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At present, brake noise is a top issue for the automotive industry having a high warranty cost if customers are dissatisfied, especially high frequency brake squeal. Despite this noise issue not impacting the performance and safety of brake system components, it affects carmaker confident and is irritating [1]. Brake squeal noise is generated when a brake system shows high vibration levels, resulting in excessively high sound pressure levels in the range of 1-16 kHz [2, 3]. The various countermeasures against high frequency brake squeal noise mechanisms have been studied by many researchers. The major theories on brake squeal mechanisms are stick-slip/sprag-slip, negative damping, and modal coupling [2, 4]. The phenomenon is characterized by self-excited oscillations caused by the friction force between brake disc and brake pads, which affects the stability of the brake system. Complex Eigenvalue Analysis (CEA) is a widely used method and tool for researchers and designers to predict the stability of brake systems. This method includes friction dynamics, the friction forces and nonlinear contact stiffness due to brake pressure stepping [1, 5, 6].

Recently, many researchers have focused on modal coupling mechanisms where out-plane and in-plane disc vibration modes are coupled together in some braking conditions to generate brake squeal noise [7]. The widely used countermeasures to prevent brake squeal high frequency noise include brake disc, brake pad structure modification to change stiffness, and material changing [8-11]. One approach to eliminate modal coupling and prevent brake squeal noise is to increase the damping of the brake system. Shims have proven to be very effective in addressing medium assembly and may alter the contact stiffness and then change the pad-caliper vibration transmissibility ratio. A positive real part dissipates vibration energy to reduce the propensity for squealing, while negative damping accumulates vibration energy [2]. There are many methods and various countermeasures to understand the high frequency brake squeal mechanism. This noise issue still impacts the automotive industry and carmakers must support high warranty costs because brake squeal is a complex phenomenon and it is difficult to clearly identify the exact root cause.

This paper studied brake squeal from high frequency noise for commercial brake systems, including a 16-inch brake disc, floating caliper and brake pads without shims on actual vehicle testing under variable conditions of braking to identify high frequency noise occurrence. Then, a numerical 3D model of the suspected brake module is established by finite element. Through instability analysis and noise mechanisms of modal coupling, the CEA numerical technique is used to identify the source. The numerical data is compared and combined with actual testing for that frequency noise phenomenon. Finally, an optimized brake pad shim is designed and modified to increase stability and structural damping to avoid the modal coupling characteristic.

2. Dynamic Instabilities and the Brake Squeal Phenomenon

High frequency brake squeal noise arises from self-excited vibration, which is under the influence of friction parameters induced while brake disc and brake pads interface during the braking condition. The solution of the steady state sliding between brake disc and brake pads may become unstable, thus contributing to dynamic instability. The way this dynamic instability sets in, in disc brakes, is referred to as the squeal mechanism [12].

2.1. Damping Effect on Brake Squeal Noise

Insight by investigation about the optimal values of structural damping that would reduce mode-coupling instability is presented, with a linear and non-linear 2-dof minimal model. This diagram is composed of a single-mass held in contact with a moving band by the mechanical elements. The contact is modelled by two plates supported by two springs, assuming that there is no separation between the plates and the band, which moves with constant velocity as in Fig. 1. Structural damping is included by introducing two dampers [12].

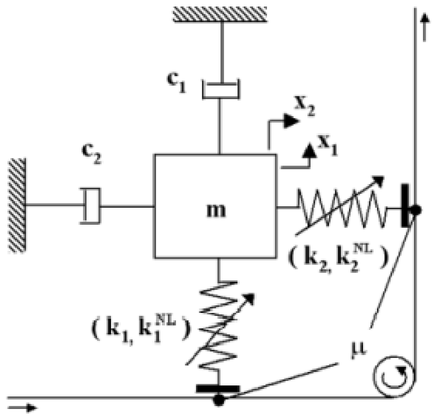

Fig. 1. Non-linear mechanical minimal model investigated [12].

Friction forces are included using Coulomb's friction law as Eq. (1).

$$F_T = \mu F_N \quad (1)$$

where,

F_T = Pull on string with force
F_N = Normal contact force

Assume that the normal contact forces are related to the displacement of the mass normal to the contact surface. The equations of motion can be written in matrix form as Eq. (2).

$$[M]\{\ddot{X}\} + [C]\{\ddot{X}\} + [K]\{X\} + \{F_{NL}(X)\} = \{0\} \quad (2)$$

where

$[M]$ = mass matrix
$[C]$ = damping matrix
$[K]$ = non-symmetric stiffness matrix
$\{X\}$ = displacement
$\{\dot{X}\}$ = velocity
$\{\ddot{X}\}$ = acceleration vectors

$\{F_{NL}(X)\}$ = the non-linear term, that depends on the cubic of displacement.

The eigenvalues of the system are found by solving the characteristic Eq. (3).

$$det(\lambda^2\,[M] + \lambda[C] + [K]) = 0 \quad (3)$$

Due to the non-symmetric nature of the stiffness matrix, some λ eigenvalues may have a positive real part, so the static solution is unstable. Friction coefficient was also taken as the control parameter in their stability analysis. [9, 12].

2.2. Modal coupling Mechanism

In brake systems, if the substructure such as brake disc, brake pads, caliper etc. have a close range of natural frequencies, the dynamical motions may couple geometrically (same wavelength) and could induce more energy into the system, thus leading to an increase in vibration amplitudes [12, 13] as in Fig. 2.

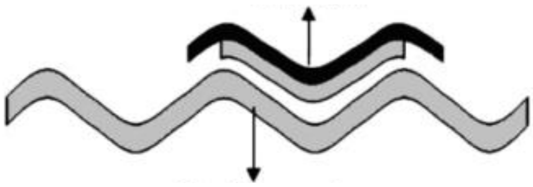

Fig. 2. Modal coupling between brake disc and brake pads [13].

Modal coupling is locked depending on operational conditions such as speed, pressure and temperature for interface characteristics such as contact stiffness, roughness, adhesive force, etc. Modal coupling of the structure involves sliding parts and the coupling results in changes of friction forces which are necessary for self-excited vibration [13].

3. Statement of Formulation and Methodology

High frequency brake squeal noise in this research is investigated and demonstrated including 3 approaches to identifying the frequency range and noise mechanism, combining the testing and numerical analysis.

3.1. Vehicle Testing

This paper studies brake noise on the commercial 16-inch ventilated brake disc and the one-piston floating caliper, which has been available for some period of time. Bruel & Kjaer type 4397 accelerometers are mounted on the floating calipers for both the left and right-side front brake modules to measure brake vibration during braking conditions as in Fig. 3. The 1/2-inch microphones are installed at the arch of the front right and left wheels as shown in Fig. 4. Then, the accelerometer and microphone data are combined and compared to confirm brake noise dominant peaks.

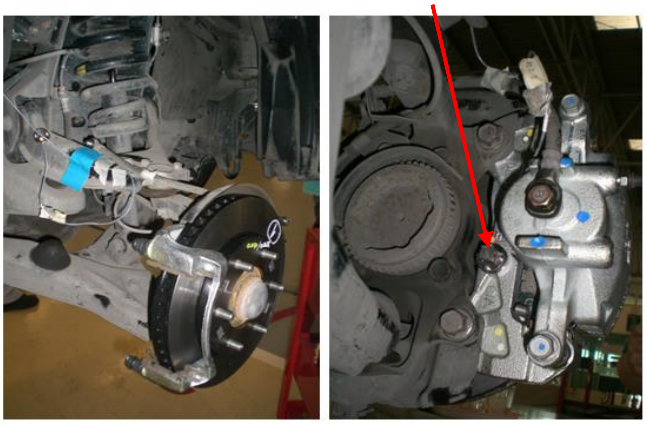

Fig. 3. Accelerometers are mounted on floating-caliper surfaces of front right and left corners.

Fig. 4. Microphones are mounted at the arch of front right and left wheels.

The boundary conditions for vehicle testing to reproduce the brake noise are found by several driving tests at an initial speed of 0-100 km/hr with braking deceleration of 0.1-0.4g. The accelerometer and microphone instruments showed the dominant peaks of vibration and brake noise corresponding are around 15 - 15.6 kHz as in the red circles of Fig. 5 and Fig. 6, respectively.

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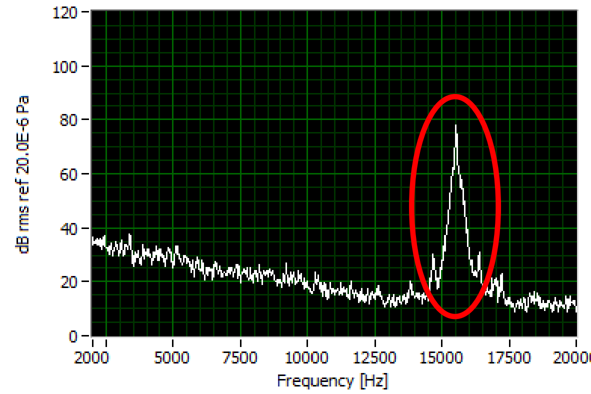


Fig. 5. Brake frequency noise spectrum by microphone detection for the front right wheel.

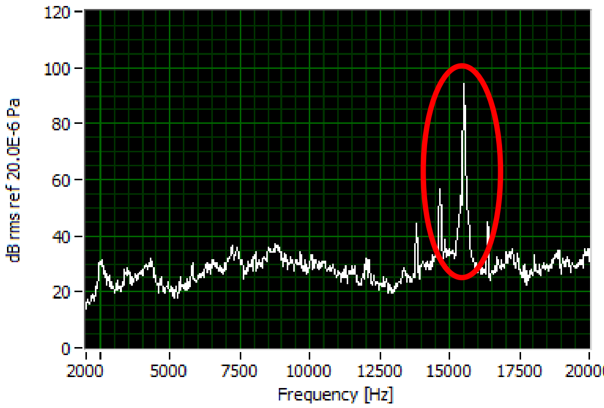


Fig. 6. Caliper vibration spectrum by accelerometer detection for the front right wheel.

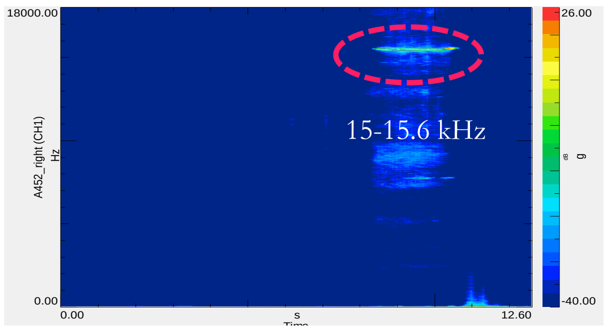


Fig. 7. Caliper vibration spectrogram by accelerometer detection for the front right wheel.

Figures 5, 6 and 7 show brake squeal noise occurrences at a temperature of 250°C, with low speed of 30 km/hr and low deceleration of 0.32g. The high frequency squeal happens mostly at low speed and the friction force tends to input (accumulate) energy into the system [11]. This phenomenon creates an instability with relative speed v(t) between brake disc and brake pads becoming negative as in Eq. (4) [14].

$$v(t) = R * \Omega(t) - V_{IPC}(2\pi f_{IPC} t) \qquad (4)$$

where

R	= the brake disc radius
$\Omega(t)$	= the brake disc displacement at time
V_{IPC}	= the brake disc in-plane circumferential (IPC) velocity
f_{IPC}	= the brake disc in-plane circumferential (IPC) frequency
t	= time (s)
π	= the ratio of a circle' circumference equal 3.14

To improve on this brake squeal noise phenomenon, the suspect brake module (brake disc, brake pads and caliper assembly) was demonstrated on a single dynamometer machine in the laboratory following the SAE J2521 standard.

3.2. Single Dynamometer Testing

To understand the parameters that affect brake squeal noise occurrence such as the friction coefficient, temperature and velocity etc., the suspect brake module is tested on a single dynamometer in the testing room, with background noise below 70 dB following the SAE J2521 standard to evaluate brake noise [15, 16].

The test results show brake squeal noise at 13.3 kHz, 15.9 kHz and 19.4 kHz under braking pressure of around 4 MPa, temperature 250 °C and a friction coefficient of around 0.4 MPa as in Fig. 8. The dominant or propensity for high frequency noise (red circle) is at 15.9 kHz, which is compatible with a vehicle testing frequency of 15.5 kHz.

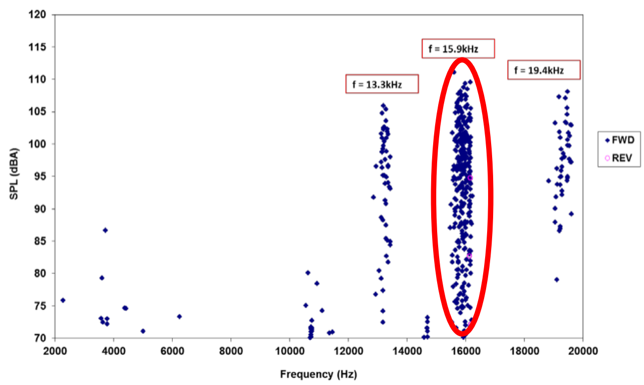


Fig. 8. The brake squeal noise result with single dynamometer testing.

This boundary condition and these results are used for reference in the simulation by FEA to identify the brake squeal noise mechanism.

3.3. Modal Testing

Modal testing is a method used to obtain dynamic characteristics such as natural frequency, mode shape, structural damping and material properties etc. of brake components on the individual part or the module [11, 16, 17]. To study and identify dynamic characteristics for a single part of the brake module in this research, the single parts, brake disc and brake pads, are supported by a sponge and accelerometers Bruel and Kjaer type 4397 installed in 3 directions on the brake disc to detect the frequency response function (FRFs) of out-plane, tangential plane and circumferential plane and 1 direction on the brake pad. The transducer hammer Bruel and Kjaer type 8204 is used for excitation as in Fig. 9 to obtains mode vibration and frequency range 4-16 kHz.

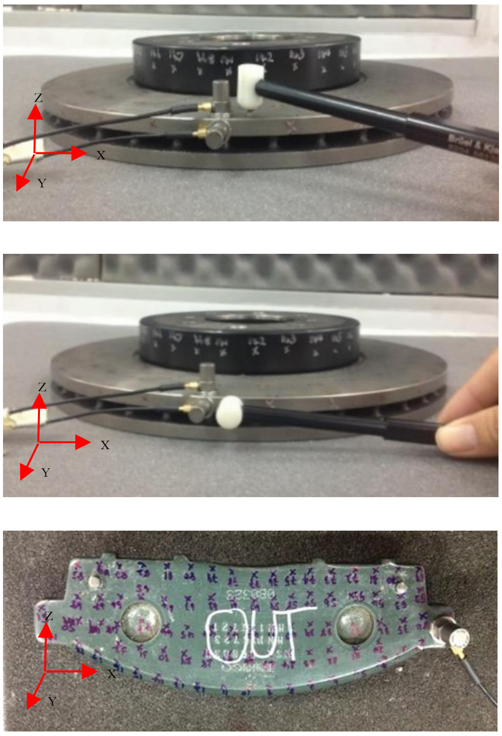


Fig. 9. The excitation and measurement points on the free-free condition for brake disc and brake pad.

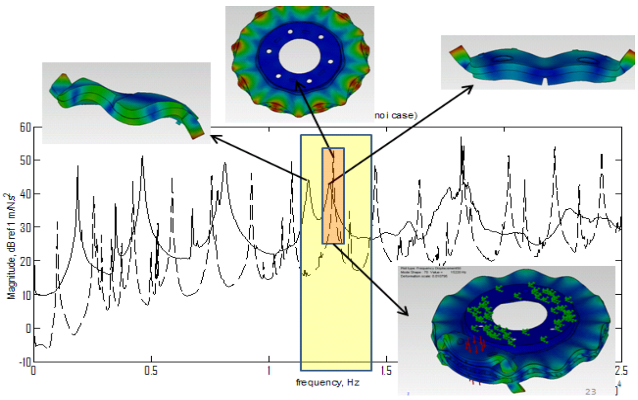


Fig. 10. Frequency response and vibration mode for brake disc and brake pads.

Figure 9 shows measurement and excitation points for brake disc and brake pads. To obtain mode vibration and frequency with the range of 4-16 kHz, brake disc and brake pads are measured at 168 and 90 points, respectively. The transducer force excites vibration modes on the brake disc in 3 directions (X, Y and Z axis) and brake pads are excited in 1 direction on the Z axis.

The modal testing showed brake disc and brake pad frequency responses coalesce at around 13 kHz and 12 kHz, respectively, as in Fig. 10. This phenomenon could grow to be mode coupling under the influence of friction coefficient excited vibration modes of the brake disc and brake pads, which have similar characteristics. To identify the noise mechanism for instability of the brake system, the CEA is used to investigate in the next section [18].

Figure 11 shows the finite element brake model for instabilities and noise mechanism simulation, and analysis by the complex eigenvalue technique. The caliper assembly is not involved due to the brake squeal high frequency noise being generated at the brake disc and brake pad interface, which arises from the friction coefficient, contact stiffness and pressure distribution [19]. Each single brake component of the finite model is numerically validated and dynamic characteristics updated by modal testing, with the deviation not exceeding 3 percent and AISI 316 stainless steel assumed for the brake pad shim as in Table 1. The FE model was then studied by instability analysis of the brake system by CEA in the next topic on the boundary condition as in Table 2.

4. Simulation and Complex Eigenvalue Analysis (CEA)

4.1. Brake Model for Finite Element Analysis

The components of the brake system including brake disc, brake pads and brake shim are established by the finite element software called SimXpert2020; the quadratic tetrahedron element type is used to model, and the thickness is divided into 2 layers for all components to study the vibration modes from 1-16 kHz. The finite element of this brake model is generated with 1,084,512 degrees of freedom (DOF) as in Fig. 11.

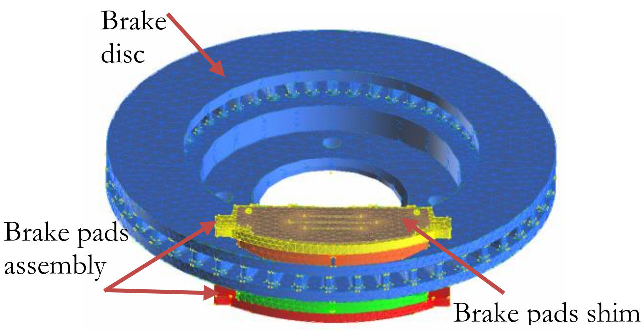


Fig. 11. Finite element model (FEM) of brake module.

Table 1. Material properties of brake components.

Item	E (GPa)	ρ (kg/m³)	Poisson's ratio
Brake disc	114	7300	0.3
Brake pads	9.5	3000	0.3
Pad black plate	200	7300	0.3
Brake pad shim	0.28	7800	0.3

Table 2. Boundary conditions for CEA analysis.

Angular Velocity (rad/s)	Friction Coefficient (Mu)	Pressure (MPa)	Structure Damping
6.28	0.5	0.5	0.005

Table 2 shows the boundary conditions for CEA analysis that were obtained from experimental testing. The angular velocity is the brake disc rotation velocity applied in the steady state condition, and the temperature parameter is not considered in this study due to the limitation of the FE software.

4.2. Complex Eigenvalue Analysis Results

This research work uses only the CEA technique to examine the stabilities of brake squeal noise in the steady state condition for design [20], by considering the negative damping value less than -0.001 and the positive real part value more than 70. The positive real part and the negative damping are identified for the unstable mode phenomenon that influences brake squeal noise.

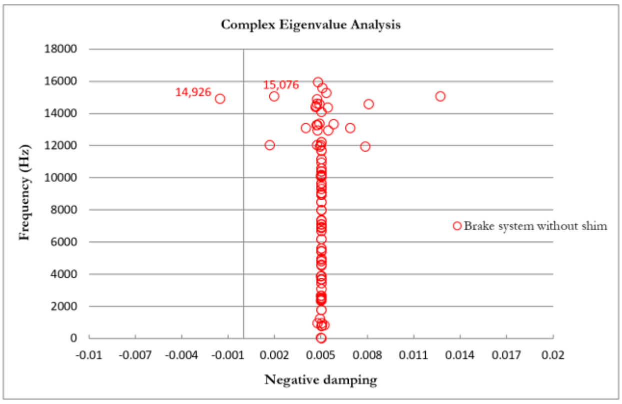


Fig. 12. Negative damping value of brake system without shim by CEA.

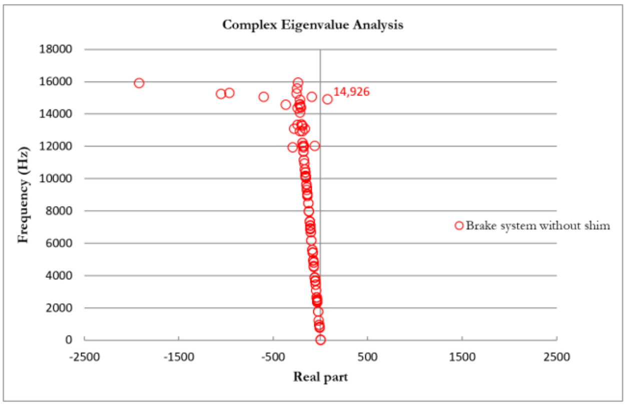


Fig. 13. Positive real part value of brake system without shim by CEA.

Figures 12 and 13 show the unstable mode results from the complex eigenvalue analysis over the frequency range of 0-16 kHz. The unstable mode generates at 1.49 kHz, where negative damping and positive real part are -0.0015 and 71.7, respectively, at the vibration mode as shown in Fig. 14.

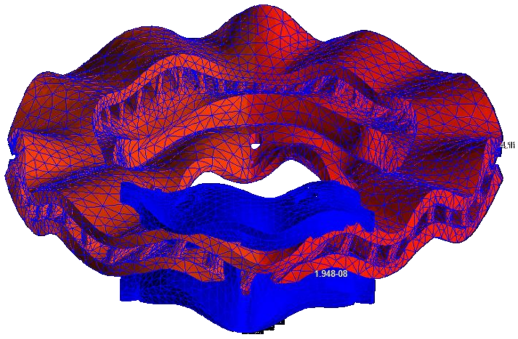


Fig. 14. Dominant unstable mode shape at frequency 14.9 kHz of brake system without shim.

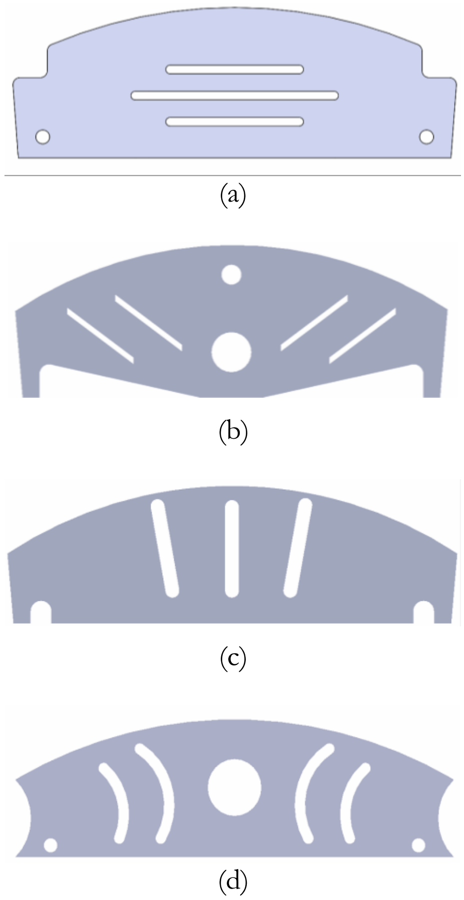


Fig. 15. Brake pads shim design1 (a), design2 (b), design3 (c) and design4 (d) configuration for CEA.

Figure 14 shows that the unstable mode shape for brake disc and brake pads are similar and correspond to the out-of-plane (bending) with the 11 nodal mode (11ND) and the 2 nodal mode (2ND) at frequency 14.9 kHz, respectively. The CEA results are in close agreement to the spectrogram of experimental testing results showed in Fig. 7.

4.3. Brake Pads' Shim Modification to Improve Unstable Modes

The aim of this research is to avoid mode coupling and increase the stability of a brake system by 4 brake pads shim configurations (design1, design2, design3 and design4) to change the stiffness as Fig. 15. The thickness of 4 shims are designed similarly at 1 mm. The brake pads, brake pads insulator/shim and brake disc stiffness change can reduce brake squeal noise [5, 21].

Figure 15 shows the brake pad shim design1 shape (a); there are 3 slots in the middle part. The brake pad shim design2 shape (b); there are 4 slots with angles of 45 degrees. The small and large circles have diameters of 6 mm and 12 mm on the center line, respectively. The brake pad shim design3 shape (c); there are 3 slots with angles 80 degrees and 90 degrees from the X-axis and the brake pad shim design4 shape (d); there are 4 curved slots and drilled holes on the middle of the shim.

4.3.1. Complex eigenvalue analysis: Brake pads shim modification

The CEA result shows 4 brake pads shim configurations (design1, design2, design3 and design4) could avoid unstable modes and there was found to be no mode coupling of brake disc and brake pads in the frequency range of 14.8-15.3 kHz as in Figures 16 and 17, and Table 3, Table 4, Table 5 and Table 6, respectively. However, the CEA results of negative damping and real part of brake pad shim design2, where the values are -0.02 and 57.3, respectively. The unstable mode of noise appears at a frequency of around 776 Hz, a low frequency moan and groan problem and does not account for the brake squeal noise. Anyway, this phenomenon could excite brake noise by the friction coefficient.

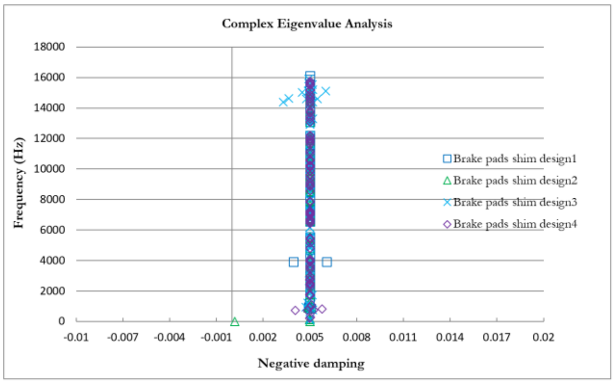


Fig. 16. Negative damping value for 4 brake pads shim modification by CEA.

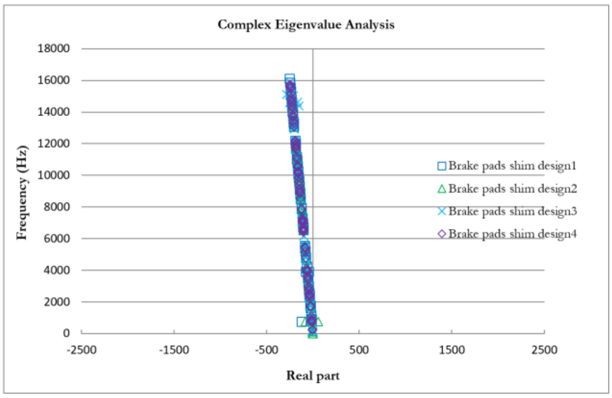


Fig. 17. Real part value for 4 brake pads shim modification by CEA.

In Table 3, the brake disc and brake pads normal modes are not coupling, brake disc vibrates in the circumferential and out-of-plane modes at frequencies of 14.8 kHz, 15.3 kHz and 15.1 kHz, respectively, and the brake pads are not deformed.

Table 4 shows the brake disc and brake pads normal modes are not coupling; the brake disc vibrates in the in-plane and out-of-plane (11 ND) modes at frequencies of 700 Hz, 14.7 kHz and 15.1 kHz, respectively and the brake pads have little deformation in the in-plane mode.

Table 5 shows the brake disc and brake pads normal modes that do not couple; the brake disc vibrates in the in-plane and out-of-plane (11 ND) modes at frequencies of 14.8 kHz, 15.2 kHz and 15.1 kHz, respectively. Brake pads have deformations in the out-of-plane mode (2ND) at a frequency of 15.2 kHz.

Table 6 shows that brake disc and brake pads normal modes are not coupling; the brake disc vibrates in the in-plane and out-of-plane (11 ND) modes at frequencies of 14.9 kHz, 15.5 kHz and 15.2 kHz, respectively. The brake pads have not deformation.

Table 3. Dominant unstable modes shape of brake system with shim design1.

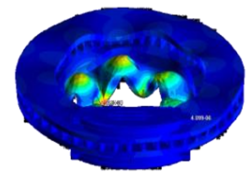
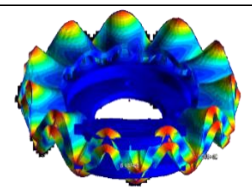
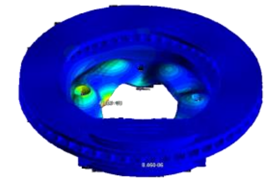
Frequency (kHz)	Real part	Negative damping	Vibration mode
14.8	-233	0.005	
15.1	-237	0.005	
15.3	-240	0.005	

Table 4. Dominant unstable modes shape at frequency of brake system with shim design2.

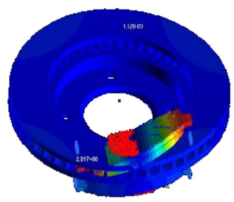
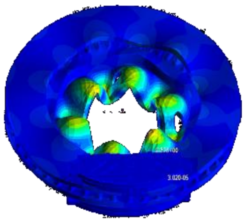
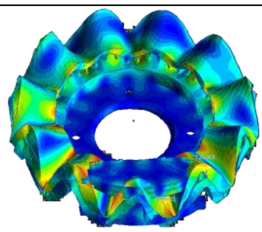
Frequency (kHz)	Real part	Negative damping	Vibration mode
0.7	57.3	-0.02	
14.7	-232	0.005	
15.1	-237	0.005	

Table 5. Dominant unstable modes shape of brake system with shim design3.

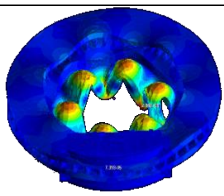
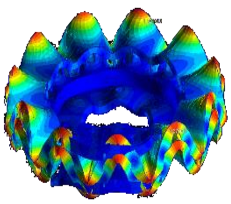
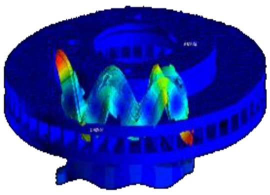
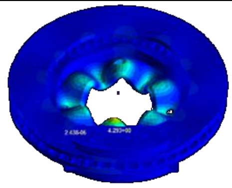
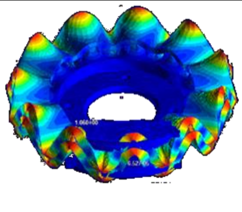
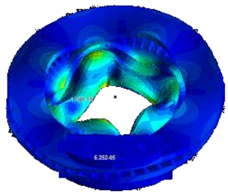
Frequency (kHz)	Real part	Negative damping	Vibration mode
14.8	-235	0.005	
15.1	-285	0.006	
15.2	-247	0.005	

Table 6. Dominant unstable modes shape of brake system with shim design4.

Frequency (kHz)	Real part	Negative damping	Vibration mode
14.9	-234	0.005	
15.2	-238	0.005	
15.5	-243	0.005	

5. Conclusion

This research studied high frequency brake squeal at 4-16 kHz for commercial brake systems. The brake pads shim is modified to increase the stiffness of the structure and avoid unstable modes. The CEA utilized the FEM of the brake disc and brake pads assembly to investigate the brake noise mechanism and study the stability of the brake system. The brake noise commercial brake system showed the behavior of the squeal mechanism which agreed well between the simulation and experimental testing results. The simulation result showed that the instability of the brake system comes from mode coupling of the brake disc and brake pads in the out-of-plane modes (11ND) and (2 ND), respectively, which arise from friction coefficient parameters. To avoid the mode vibration in out-of-plane, a brake pad shim was used to modify the brake pads to increase the stiffness with 4 designs. The CEA showed that brake pad shim modification on the brake pads could avoid the out-of-plane mode (2ND) vibration efficiently for the high frequency brake squeal noise (HFBS) [2, 16, 20, 21]. Even these structure modifications were able to avoid HFBS but there were some brake pad shim makes the noise move toward the direction of the lower frequency such as brake pad shim design1, design2 and design3, respectively. For this study, the brake pad shim design4 was able to avoid high frequency brake squeal and low frequency noise efficiency and was the optimum due to the maximum real part value being -234, which is higher than the others.

This approach studied one unique commercial brake system for a 16-inch brake disc and steady state conditions, which only accounts for friction coefficient, pressure and velocity due to the limitation of facilities. Therefore, to exactly understand the brake squeal noise mechanism the temperature parameter and transient dynamic analysis are the main factors to study in the future.

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References

[1] V. Kumar, S. V. Chaitanya, and S. B. Phadke, "Automotive brake disc design to suppress high frequency brake squeal noise," *International Journal of Engineering Research & Technology (IJERT)*, vol. 3, no. 1, pp. 165-169, 2014.

[2] J. Glisovic and D. M. Miloradovic, "Eliminating brake noise problem," *Mobility & Vehicle Mechanics*, vol. 36, no. 3, pp. 35-51, 2010, Art. no. UDC: 629.113592.12: 534.322

[3] A. M. George, R. S, and Nithin Prasad, "Squeal analysis of disc brake rotor using finite element approach," *International Journal of Engineering Research & Technology*, vol. 6, no. 02, pp. 349-354, Feb. 2017.

[4] J. Park, "The relationship between squeal noise and stick-slip vibration," Mando Corporation, South Korea, May 2018.

[5] D. Schmid, V. Sessner, N. Grabner, U. von Wagner and K. A. Weidenmann, "Parameter identification of brake pad shims for complex eigenvalue analysis," in *PAMM. Proc. Appl. Math. Mech*, vol. 19, no. 1, p. e201900306, May 2019.

[6] M. Esgandari, "Simulation methods for vehicle disc brake noise, vibration & harshness," Ph.D. thesis, The University of Birmingham, Aug. 2014.

[7] A. Khuntiptong and C. Chantalakhana, "On the study of disc cheek thickness as a design parameter for reducing brake squeal noise," in *Proceedings of the 6th TSME International Conference on Mechanical Engineering*, 16-18 December 2015.

[8] G. Pan and L. Chen, "Impact analysis of brake pad backplate structure and friction lining material on disc-brake noise," *Advances in Materials Science and Engineering*, vol. 2018, 2018, Art. no. 7093978.

[9] S. Khuntiptong, C. Chantalakhana, S. Sucharitpwatskul, T. Luengsrisakul, and C. Sedkit, "On the study of in-plane and out-plane rotor vibration modes effected to brake squeal noise," in *Proceedings of the 27th Conference of the Mechanical Engineering Network of Thailand*, 16-18 October 2013.

[10] A. Tuchinda, "Development of validated models for brake squeal predictions," Ph.D. thesis, The University of London, Sep. 2003.

[11] A. Khuntiptong and C. Chantalakhana, "Stability improvement of brake disc to modal coupling at high frequency squeal," *Applied Science and Engineering Progress,* vol. 15, no. 1, pp. 1-11, 2022.

[12] A. J. da Guia Rodrigues, "Disc brake squeal modal coupling instability type," master's thesis, Laboratory of Vibrations of Mechanical Systems, The University of Porto, 2017.

[13] N. M. Ghazaly, I. Ahmed, and M. R. EI-Sharkawy, "A review of automotive brake squeal mechanisms," *Journal of Mechanical Design and Vibration*, vol. 1, no. 1, pp. 5-9, Jan. 2014.

[14] M. Yang and A.-H. Afaneh, "A study of disc brake high frequency squeal and disc in- plane/out-of-plane modes," Robert Bosch Corporation, Farmington Hills, Michigan, USA, May 2004.

[15] *Standard of Automotive Engineering*, SAE J2521, 2003.

[16] S. A. Rao, M. A. Jusoh, A. R. A. Baka, M. K. A. Hamid, and K. Zakaria, "Reduction of disc brake squeal noise using constrained layer dampers," *Jurnal Teknologi*, vol. 79, pp. 83-87, Dec. 2017.

[17] A. Belhocine and N. M. Ghazaly, "Effect of young's modulus on disc brake squeal using finite element analysis," *International Journal of Acoustic and Vibration*, vol. 21, no. 3, pp. 292-300, Sep. 2016.

[18] S. Yeamdee, T. Bovornratanaraks, M. Sriwiboon, C. Chantalakhana, and C. Ratanasumawong, "Effect of slot and chamfer shape of brake pad on mode coupling," SAE International, Mar. 2019.

[19] T. Matsushima, K. Izui, and S. Nishiwaki, "Concept design method for reducing brake squeal in disk brake systems considering," Journal of Mechanical Design, vol. 134, p. 061008, 2012.

[20] M. Esgandari and O. Olatunbosum, "Computer aided engineering prediction of brake noise: Modeling of brake shims," *Journal of Vibration and Control*, vol. 22, no. 10, pp. 2347-2355, 2014.

[21] G. Pan and L. Chen, "Impact analysis and optimization of material parameters of insulator on brake squeal," *IEEE Access*, vol. 7, pp. 15861-15867, 2019.

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